DTU Library

Screening of heat pump performance improvements obtained through gradual heating using a tank system

Kofler, René; Elmegaard, Brian; Madsen, Claus; Olsen, Lars; Markussen, Wiebke B.

Published in:
Proceedings of the 25th IIR International Congress of Refrigeration

Link to article, DOI:
10.18462/iir.icr.2019.1055

Publication date:
2019

Document Version
Publisher's PDF, also known as Version of record

Citation (APA):
Kofler, R., Elmegaard, B., Madsen, C., Olsen, L., & Markussen, W. B. (2019). Screening of heat pump performance improvements obtained through gradual heating using a tank system. In *Proceedings of the 25th IIR International Congress of Refrigeration* (pp. 4608-4615). International Institute of Refrigeration. https://doi.org/10.18462/iir.icr.2019.1055

Manuscript ID: 1055
DOI: 10.18462/iir.icr.2019.1055

Screening of heat pump performance improvements obtained through gradual heating using a tank system

René KOFLER[(a)], Brian ELMEGAARD[(a)], Claus MADSEN[(b)], Lars OLSEN[(b)], Wiebke B. MARKUSSEN[(a)]

[(a)] Technical University of Denmark, Kgs. Lyngby, 2800, Denmark, renekof@mek.dtu.dk
[(b)] Danish Technological Institute, Taastrup, 2630, Denmark, clma@teknologisk.dk

ABSTRACT

Many industrial processes require both cooling and heating of process streams. The use of a co-generation heat pump system enables the simultaneous production of both utilities within one single, electrically driven unit.

Within this work, the addition of a tank system at the hot end of different heat pump systems was investigated. The tank system consists of a stratified storage tank for the storage of hot water and a smaller charge tank, which allows for multiple circulations of the water through the condenser. This leads to a gradual heating of the water, leading to a higher COP compared to heating in one step.

Using numerical modelling and simulation, a screening was carried out investigating the influence of different working fluids and heat pump configurations, different temperature glides in the heat sink and different temperature lifts on the performance of the heat pump system. The largest improvement, compared to a conventional heat pump, was achieved with a one-stage heat pump with R600a, increasing the COP by 35 %.

Keywords: Heat pump, COP, Energy Efficiency, Working Fluid, Tank.

1. INTRODUCTION

Many countries aim to reduce the use of fossil fuels. The long-term Danish energy policy aims for a fossil fuel free society by 2050. This includes all sectors, including the heating and transport sector. Bühler et al. (2016) conducted an energy and exergy analysis of the Danish industry sector. The results show potentials for reduction of fossil fuel use and exergy losses through waste heat recovery and the use of alternative technologies.

Heat pumps (HPs) allow for an efficient, fossil fuel free production of heat, when using electricity from renewable energy carriers like wind and solar. Additionally, HPs allow for process integration by using waste heat streams, which have to be cooled down, while heating another stream at a higher temperature level. The cooled stream could be a warm stream, which is cooled down to ambient conditions before releasing it to the environment. It could also be a stream used for cooling purposes within the same process.

Units, providing heating and cooling capacity are referred to as co-generation HPs. The efficiency of such units is limited due to the high temperature lift between the cold and hot process water. In order to improve the efficiency of these systems, different cycle configurations, working fluids or new system designs can be used. One interesting approach is serial or gradual heating of water. Gradual heating leads to lower condensing temperatures on average and hence to higher coefficient of performance (COP) compared to heating in one step with conventional heat pumps.

Wang et al. (2010) conducted a comparison between a one-stage cycle, a two-stage cycle with intercooler and serial heating by two serial, one-stage HPs. The results show that below a specific temperature lift, the two-stage cycle achieves the highest COP, while the a configuration with two HPs in series is the most efficient above the threshold. Ommen et al. (2015) investigated the technical and economic feasibility of serially connected industrial HPs. It is shown that serial connection of two or three HPs leads to higher COPs and lower net present values (NPV) compared to one single HP.

Serial heating can also be realized by using storage tanks instead of several HPs in series. This approach is presented by Rothuizen et al. (2014) and Olsen et al. (2015) termed the ISEC concept and by Löffler and Griessbaum (2014) termed the trapezoid cycle. In both concepts, two storage tanks are used, where one tank is charged gradually, until it is fully charged with water of the wanted

supply temperature. In the meanwhile, the second tank is discharged, delivering hot water to the process from the top of the tank, while colder return water is filled into the bottom of the discharge tank.
The ISEC concept shows an improvement potential of up to 25 % in COP compared to a conventional HP, where a one-stage HP working with ammonia (R717) was used to heat water from 40 °C to 80 °C with an evaporation temperature of 22 °C (Rothuizen et al., 2014). The ISEC concept was furthermore tested experimentally, investigating the performance of the HP and the stratification in the storage tanks (Rothuizen et al., 2015 and Olsen et al., 2016).
Experimental investigations of the trapezoid cycle show COP improvements of 10 % - 50 % for water inlet temperatures at the condenser between 12 °C – 37 °C. The supply temperature of the hot water and the inlet temperature to the evaporator were kept constant at 42 °C and 10 °C, respectively. A one-stage HP using R134a was used for the experiments (Löffler and Griessbaum, 2014).
In this work, a system similar to the described systems using storage tanks was investigated. The main difference of the new system to the ones proposed by Rothuizen (2014) and Löffler (2014) was a different setup of the storage tanks.
In the previous works investigating gradual heating, only specific cases using specific HP configurations, working fluids, temperature sets etc. are considered. The aim of the screening is to reveal which of the investigated parameters mainly influence the possible performance improvements through gradual heating, and to identify the most promising system setups and system boundaries. The screening considered different heat pump configurations, different working fluids and different temperature sets.

2. METHODS

In the following sections, the new design is introduced, different HP configurations used for the screening are shown, the modelling strategy and the models of the different components of the systems are described and the assumptions for the different screening cases are explained.

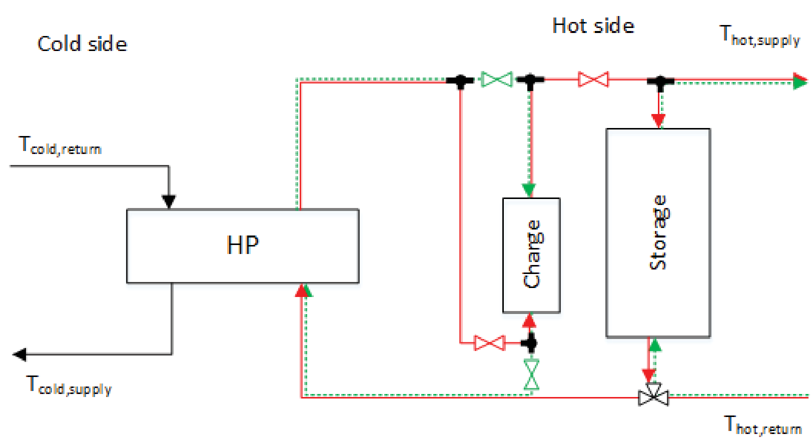

Figure 1: Principle sketch of the system. The green dashed line at the hot side denotes the flow during the charging process of the charge tank and the red solid line the one during discharging

2.1. The Concept

The main novelty of the proposed system is the tank setup, shown in Figure 1. Instead of using two tanks of the same size, one smaller charging tank and a bigger storage tank are used. This setup introduces two advantages compared to the previously described systems with two charging tanks of the same size. Firstly, the new system requires less three-way valves than the ISEC concept, which will reduce the investment costs. Secondly, many existing HP systems for industrial processes already use buffer tanks in order to compensate for varying heat loads over time. These tanks can be used as storage tanks for the new system. This allows for increasing the efficiency of existing HP systems by adding only one smaller charge tank, reducing the investment costs compared to the ISEC concept.
In Figure 1, the two different operation modes of the tank system at the hot side are depicted. The green dashed lines show the water flow during the charging process of the charge tank. In this

period, colder water is taken from the bottom of the charge tank, heated in the condenser of the HP and the hot water is returned to the top of the charge tank. In the meanwhile, hot process water at the required supply temperature is delivered to the process from the top of the stratified storage tank. The cold water returning from the process is fed to the storage tank at the bottom in order to insure stratification within the tank. When the charge tank is filled completely with water at the required supply temperature, the operation mode is switched to discharge the charge tank. This is shown in Figure 1 with red solid lines. In this mode, the return water from the process flows together with water from the bottom of the storage tank to the condenser of the HP. The heated water is then fed to the bottom of the charge tank. This leads through the stratification within the charge tank that the hot water at the top is supplied to the process. A part of the flow goes into the storage tank in order to refill it. When all the water at supply temperature has left the charge tank, the charging process of the charge tank starts again.

2.2. Heat Pump Configurations

This work aims at the investigation of the influence of different factors on the improvement potential though gradual heating. One of these factors is the HP configuration. In Figure 2, the investigated HP configurations are depicted.

In Figure 2, left, a simple one-stage cycle consisting of an evaporator, compressor, condenser and throttling valve is shown. The condenser is divided into two separated heat exchangers for desuperheating and condensation. This division is connected to the modelling approach, as explained in the following sections. The technical realization can be both, in two separate heat exchangers or one component for the whole process.

Figure 2, middle shows a one-stage cycle with regeneration heat exchanger (RHE), often called liquid-suction or internal heat exchanger. This component is often used for refrigeration or heat pump cycles with HFCs, hydrocarbons or similar working fluids.

A two-stage cycle with open intercooler is depicted in Figure 2, right. This setup is commonly used for refrigeration or heat pump cycles with large temperature lifts. It allows for an increase in efficiency and for higher pressure ratios between the evaporation and condensation pressure.

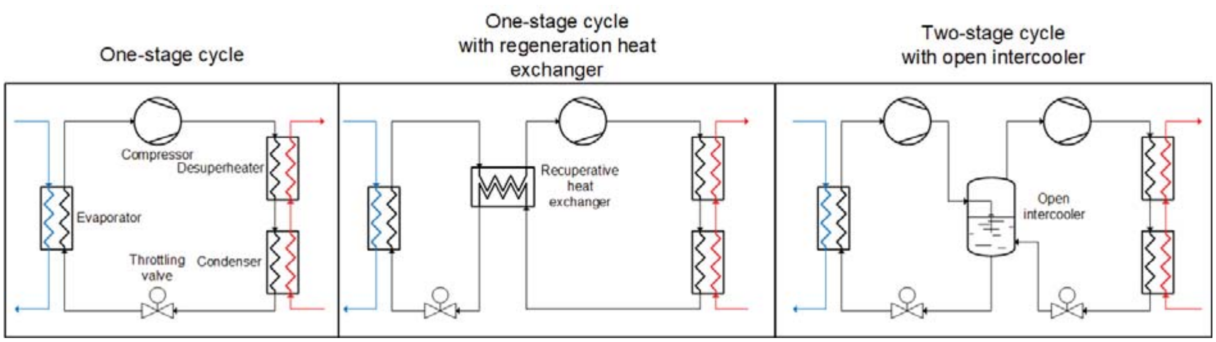

Figure 2: Different heat pump configurations examined in this work. Left: simple one-stage cycle. Middle: one-stage cycle with regeneration heat exchanger. Right: two-stage cycle with open intercooler

2.3. Modelling Strategy

For this work, a robust model for quick calculations of different cases was built. With the system, water was heated gradually in N circulations from a defined return temperature to a defined supply temperature. From a model-point-of-view, the first circulation was seen as a design circulation. For the first circulation, the minimum temperature differences in the evaporator and the condenser and quality of the working fluid between the condenser and desuperheater were defined. From these, the UA-values of the three heat exchangers were derived and kept constant for the following circulations. The UA-value of the RHE was determined by fixing the heat exchanger effectiveness during the first circulation.

The compressors were working at constant displacement during all circulations and the water mass flow through the condenser and desuperheater was also kept constant. The constant water mass flow rate allowed for an equal time period of each circulation. For the energy balance of the HP, one degree of freedom is required for the heat input from the heat source at the cold side. Here the return and supply temperatures were set constant, leading to the water mass flow to be variable between the different circulations. The ratio of displacement between the low- and high-stage compressor for the two-stage cycle was determined by defining the intermediate pressure at the intercooler with the geometrical mean value, calculated as the square root of the product of evaporation and condensation pressure.

2.4. Component Models

As shown in section 2.2, all HP cycles consist of the following components: compressors, heat exchangers, throttling valves, intercooler (if applicable). All components were modelled based on steady state mass and energy balances. Heat losses in all components were neglected. The compressors were modelled by volumetric and isentropic efficiency. For both efficiencies, polynomials in dependency of the evaporation and condensation temperatures were used. The throttling valves were modelled as isenthalpic processes. The heat exchanger models were based on the logarithmic mean temperature difference (LMTD) and a constant UA-value. In order to calculate the UA-value of the evaporator and the condenser, the minimum temperature difference between water and working fluid within the heat exchanger was fixed. In the evaporator, the minimum temperature difference was found to be between the outlet temperature of the water and the evaporation temperature of the working fluid. The condenser was divided into two heat exchangers, one for desuperheating of the working fluid and one for condensation, as shown in Figure 2. A UA-value for each of the two heat exchangers was determined. The minimum temperature difference was found to be between the condenser and the desuperheater. Additionally, the inlet quality to the condenser was fixed. In this work, no subcooling of the working fluid within the condenser was included. The size of the RHE was determined by fixing the heat exchanger effectiveness, which describes the ratio between the transferred heat over the theoretical maximum heat transfer. The open intercooler was completely determined through the mass and energy balances. Complete interstage cooling within the intercooler was assumed, meaning that the outlet quality of the vapour going to the high-stage compressor was set to one.

The tank system was modelled by quasi-static mass and energy balances. Further, we assumed both - charging and storage tank, to be ideally stratified, i.e. without mixing of layers and heat transfer between layers and through the tank walls. In order to account for the increasing energy consumption of the water pump with increasing water mass flows, a simple pump model based on a constant pressure loss within the condenser, the water volume flow and a constant isentropic efficiency was used.

2.5. Test Cases for Screening

In this work, different screenings were conducted. The first screening investigated the performance of the system for different working fluids and cycle configurations for different numbers of circulations per charging cycle of the charging tank. For this screening, cold water was cooled from 20 °C to 10 °C while heating water from 20 °C to 60 °C at the hot side. The number of circulations was varied from 2-15 and then compared to the direct heating HP without a tank system. The examined working fluids were R134a, R290, R600a and R717. For all working fluids, calculations with a simple one-stage HP were conducted. For R717, a two-stage cycle with open intercooler and for the other working fluids, a one-stage cycle with RHE were investigated additionally.

In the next screening, the influence of the temperature glide in the heat sink was examined. The temperature glide in the heat sink is defined as the difference between the return and supply temperature on the hot side. For the variation, the supply temperature was held constant at 60 °C, while the return temperature was varied from 20 °C to 55 °C. The temperature of the heat source was kept constant.

Finally, the influence of the temperature lift was examined by varying the hot supply temperature between 40 °C and 80 °C while keeping the temperature glide in the heat sink at 20 K. The temperature lift of the process is defined as the difference between the supply temperature and the return temperature of the cold water. By keeping the cold side at the supply temperature of 10 °C and a return temperature of 20 °C, temperature lifts of 20 K to 60 K were investigated.

For the cases using ammonia (R717) as a working fluid a flooded evaporator was modelled, assuming a vapour quality of x=1 at the inlet of the (low-stage) compressor. For other working fluids, a direct expansion evaporator with a compressor suction superheat of 5 K was modelled. The operating limits for the compressors were defined with maximum compressor discharge temperatures of 180 °C (Ommen et al., 2015).

In the design run of the evaporator and the condenser, pinch point temperature differences of 5 K were used. The RHE effectiveness was fixed to 0.6. For the calculation of the power consumption of the hot water pump, a pressure drop in the condenser of 0.5 bar and an isentropic efficiency for the pump of 0.8 were assumed.

For the cycles working with R717, compressor data from Sabroe were used. For the other working fluids, a compressor from GEA Bock were used. The names of the compressors are shown in Table 1 together with the previous explained assumptions. The data for the fitting of the compressor polynomials were taken from Johnson Controls Denmark (1985-2019) and IPU (2008-2015) for the Sabroe- and Bock-compressors, respectively.

Table 1: Fixed input variables for heat pumps and water pump

Working fluid	R717	R134a, R290, R600a
Compressor suction superheat [K]	0	5
Max. discharge temperature [°C]	180	180
Pinch point temperature difference [K]	5	5
Regeneration heat exchanger effectiveness [-]	-	0.6
Water pressure drop in condenser [bar]	0.5	0.5
Pump isentropic efficiency [-]	0.8	0.8
Compressor model (one- and high-stage)	Sabroe HPX-708	Bock EX-HG 12P/60 4S
Compressor model (low-stage)	Sabroe SMC-112L	-

3. RESULTS

This section presents the screening of the HP system using storage tanks for gradual heating. The boundary conditions for the screening were presented in the previous section.

3.1. Working fluid and heat pump configuration

In the first screening, the influence of the working fluid and the HP configuration on the COP of the system was examined. Figure 3, right, shows the COP of the conventional HP as coloured columns for different working fluids and HP configurations. As seen, changing the HP configuration from one-stage to two-stage or including a RHE leads to an improvement in COP for each working fluid. The boxes on top of the columns show the range of COPs, when using the tank system for gradual heating with different number of circulations of water through the charge tank until it is completely charged.

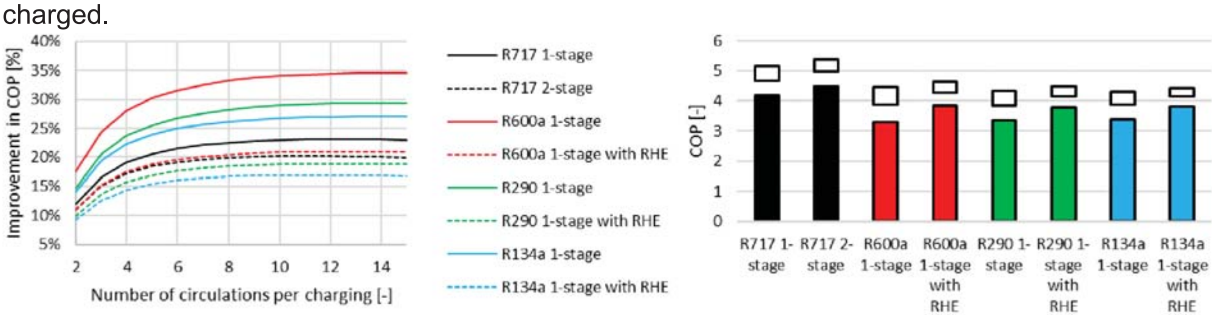

Figure 3: Left: Improvement in COP compared to conventional heat pump for different working fluids and heat pump cycles. Right: Coloured column depicts COP of conventional heat pump, box on top the range for different numbers of circulations during charging

Figure 3, left, shows the improvement of COP compared to a conventional HP for different working fluids and HP configurations and different numbers of circulations per charging cycle of the charge tank. The solid lines denote the improvement when using a simple one-stage HP and the dashed lines for two-stage HPs or with a RHE. It may be seen that the use of the tank system leads to bigger improvements for the simple one-stage HP configuration than for the other configurations. The difference in improvement between one-stage and two-stage HPs is smaller than between the one-stage HPs with and without RHE. It is also visible, that different working fluids show different behaviour. The biggest improvement in COP was achieved for a one-stage HP with R600a, with an improvement of up to 35 % and a minimum improvement for two circulations per charging cycle of 18 %.

3.2. Temperature Glide in Heat Sink

As a second parameter, the temperature glide in the heat sink was examined. Figure 4, left, depicts the maximum improvement in COP compared to a conventional HP for different temperature glides in the heat sink, working fluids and HP configurations. The values for a temperature glide of 40 K correspond to the maxima of the curves in Figure 3, left. It may be seen that the maximum improvement decreases with decreasing temperature, reaching an improvement of only 1.2 % for a one-stage HP with R600a and no improvement for a temperature glide of 5 K for a two-stage HP or RHE with R717 or R290 and R134a respectively.

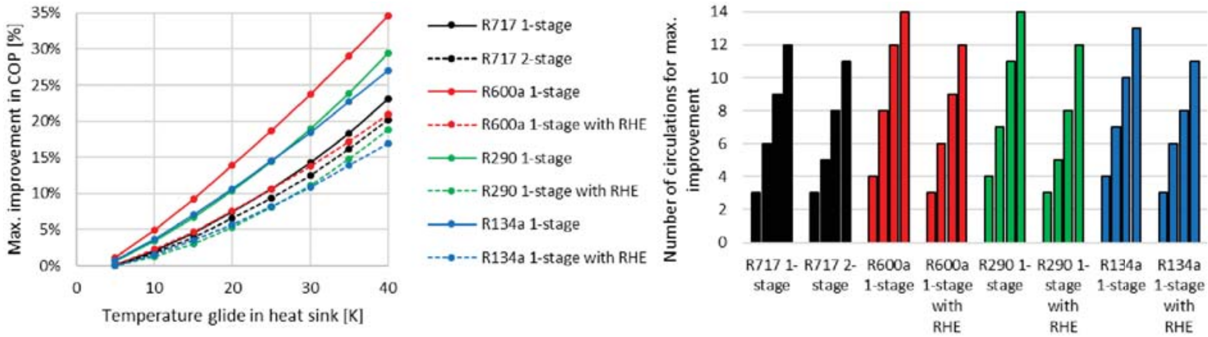

Figure 4: Left: Improvement in COP for varying temperature glide in heat sink. Right: Number of circulations with maximum COP for temperature glides of 10, 20, 30 and 40 K.

Figure 4, right, shows the optimum number of circulations per charging cycle for temperature glides of 10 K to 40 K in steps of 10 K. The fourth column of each working fluid and HP configuration shows the location of the maximum in Figure 3, left. It may be seen that the location of the maximum moves to lower numbers of circulations for decreasing temperature glides.

3.3. Temperature lift

The final screening investigates the influence of the temperature lift on the performance of the novel HP system. Figure 5, left, shows the maximum improvement in COP for different temperature lifts of 20 K to 60 K. The results with a temperature lift of 40 K coincide with the data for a temperature glide in the heat sink of 20 K in Figure 4, left. The data for the one-stage HP with R717 are shown only for temperature lifts of 20 K to 40 K. For higher temperature lifts, the discharge temperature exceeds the maximum allowed temperature of 180 °C. The highest improvement in COP was found for a one-stage cycle with R290 at a temperature lift of 20 K with 17 %. For temperature lifts above 20 K, the biggest improvement is achieved with a one-stage with R600a.

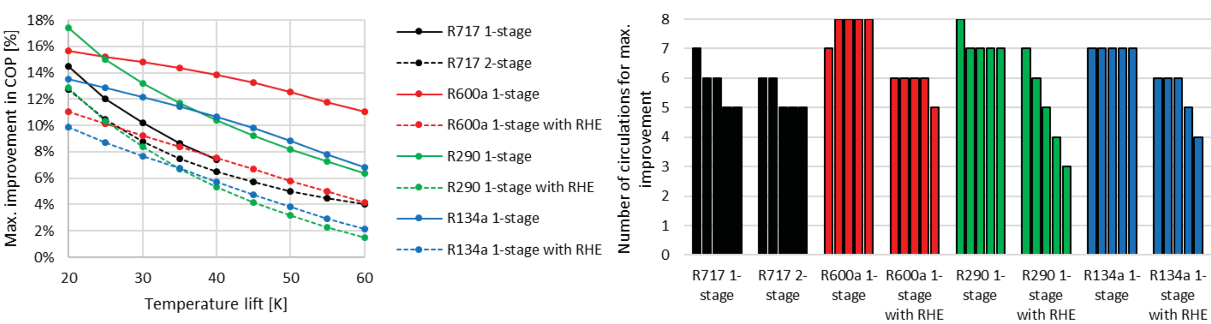

Figure 5: Left: Improvement in COP for varying temperature lifts. Right: Number of circulations with maximum COP for temperature lifts of 20, 30, 40, 50 and 60 K

For all investigated cases, the increase in temperature lift leads to lower maximum improvement. However, the slope of the curves shown in Figure 5, left, differs between the cases. While the curves of R717 and R290 are steeper at lower temperature lifts for all HP configurations, they are close to linear for R600a and R134a in a one-stage HP with RHE and for simple HP the get steeper with increasing temperature lifts.
Figure 5, right, shows the location of the maximum for temperature lifts between 20 K to 60 K in steps of 10 K. It shows some differences between the cases. For a one-stage cycle with R600a, the

number of circulations for the maximum improvement in COP increases with increasing temperature lifts. For a one-stage cycle with R134a, it remains constant at seven circulations and for all other cases it decreases with increasing temperature lifts.
Figure 6 shows the COP at different temperature lifts of a one-stage and two-stage cycle with R717 for a conventional HP and the novel system using storage tanks. As in Figure 5, left, the data for the one-stage cycle are shown only for temperature lifts of 40 K or lower, due to the limitations in the discharge temperature. As expected, all COPs decrease with increasing temperature lifts and the COP of a conventional one-stage HP is lower than the one of a conventional two-stage HP. As seen in the previous screenings and also in Figure 5, left, the introduction of the tank system leads to a bigger improvement for the one-stage cycle than for the two-stage cycle. In this case, this leads to higher COPs for a one-stage configuration with storage tanks than for a conventional two-stage HP for temperature lifts below 35 K. For a temperature lift of 20 K, the COP of the one-stage cycle with tanks of 6.48 reaches almost the COP of the two-stage HP with tanks of 6.66.

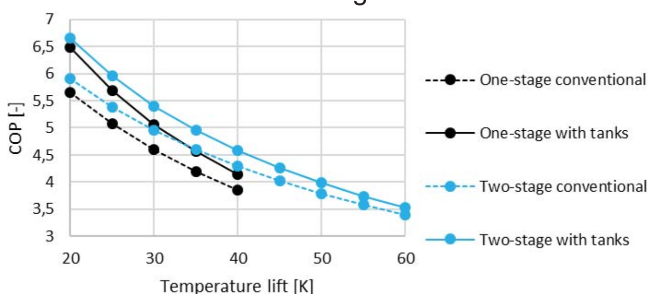

Figure 6: COP of one-stage and two-stage cycles with and without tank system with R717 for different temperature lifts

4. DISCUSSION

The conducted screenings showed the improvement potential of the novel storage system for different working fluids and HP configurations. However, the numerical results depend on the assumptions made for the calculations. A big influence derives from the choice of the compressors. Choosing another compressor would lead to different isentropic and volumetric efficiencies and thus for different COPs. Another influential factor are the assumptions for the electricity consumption of the water pump, i.e. the pressure drop in the condenser and the pump isentropic efficiency. Changing these values would change the maximum achievable improvement and the optimum number of circulation per charging cycle.
The assumption of constant UA-values of all heat exchangers after the design run was made for increasing the robustness and speed of the model. However, the varying mass flow rates of the working fluid and the water on the cold side could have an influence on the heat transfer coefficients and thereby on the performance. The use of the minimum temperature difference or the heat exchanger effectiveness as a design variable for the heat exchangers means that different heat exchanger sizes were considered for the screenings. Hence, the screening compared newly built HP systems to each other. For the upgrading of existing system, fixing the size of the heat exchangers would be more realistic and lead to different results than the ones presented in this work. It should be mentioned, that the assumption of ideally stratified storage tanks and the neglect of dynamics of the HP means that in this work a best-case scenario in terms of COP was examined. In future work, the influence of thermal inertia and the control of the system should be investigated.
Within this work, only gradual heating with a tank system on the hot side was examined. Ommen et al. (2015) showed that the best performances for systems with serial heating are achieved, when serial heating and cooling is conducted at the same time. This corresponds to using the tank system at both hot and cold side. However, when using the tank system instead of serially connected heat pumps, the control and operation of the system becomes very complex. Further investigation on the potential of gradual heating and cooling and on the operation and control of such systems by dynamic models and experiments will be necessary.

5. CONCLUSION

The novel system, using a small charging tank and a bigger storage tank for gradual heating of process water showed significant potential for reduction in energy consumption. For the investigated

temperature sets, a one-stage heat pump with R600a as working fluid showed the biggest potential of improvement of all cases. When cooling water from 20 °C to 10 °C and heating water from 20 °C to 60 °C in 14 steps, the COP improvement was 35 % compared to a conventional heat pump, heating the water in one single step. The results showed that the improvement potential in COP decreases, when the heat pump configuration is changed from a simple one-stage cycle to a two-stage cycle or by adding a regeneration heat exchanger.
For all cases, a reduction of the temperature glide in the heat sinks leads to smaller improvement in COP. The trend of the maximum improvement showed to be independent from the working fluid and heat pump configuration. For some cases, the addition of the storage tanks does not lead to an improvement in COP at very low temperature glides.
Increasing the temperature lift between heat source and heat sink showed a decreasing improvement potential. The behaviour of the maximum improvement with increasing temperature lift differs between different working fluids. Further investigations on the reasons for these different behaviours will be necessary.
For R717, it was shown that for temperature lifts below 35 K, higher COPs can be achieved with a one-stage heat pump with storage tanks than with a two-stage cycle without storage tanks. For a temperature lift of 20 K, the tank system achieves similar COPs for one-stage and two-stage heat pumps.

ACKNOWLEDGEMENTS

This research project is financially funded by EUDP (Energy Technology Development and Demonstration). Project title: Highly Efficient and Simplified Thermodynamic Cycle with isolated heating and cooling – cost optimized (ISECOP), project number: 64017-05102.

REFERENCES

Bühler F., Nguyen T-V., Elmegaard B., 2016. Energy and exergy analyses of the Danish industry sector. Applied Energy, 184, 1447-1459.

F-Chart Software (LLC.), 1992-2018. Engineering Equation Solver,EES. V10.478 (21/8/2018). http://www.fchart.com/ees/. [accessed 10.01.2019]

IPU, 2008-2015. Pack Calculation Pro. V4.20.

Johnson Controls Denmark, 1985-2019. Sabroe MatchMaster Program COMP1. V26.70.

Löffler M. K., Griessbaum N., 2014. Storage devices for heat exchangers with phase change. Int. J. Refrigeration, 44, 189-196.

Olsen, L. et al., 2015. High efficient heat pump system using storage tanks to increase cop by means of the ISEC concept (part II, thermal storage system). Proceedings of the 24th IIR International Conference of Refrigeration, International Institute of Refrigeration.

Olsen, L. et al., 2016. Highly Efficient Thermodynamic Cycle with Isolated System Energy Charging (ISEC) – Final Report. Danish Technological Institute.

Ommen T. et al., 2015. Enhanced technical and economic working domains of industrial heat pumps operated in series. Proceedings of the 24th IIR International Conference of Refrigeration, International Institute of Refrigeration.

Rothuizen E. et al., 2014. High efficient ammonia heat pump system for industrial process water using the ISEC concept – part 1. Proceedings of the 11th Gustav Lorentzen Conference on Natural Refrigerants, Chinese Association of Refrigeration.

Rothuizen E. et al., 2015. High efficient heat pump system using storage tanks to increase COP by means of the ISEC concept – part 1. Proceedings of the 24th IIR International Conference of Refrigeration, International Institute of Refrigeration.

Wang K. et al., 2010. Investigation of the performance of a high-temperature heat pump using parallel cycles with serial heating on the water side. Int. J. Refrigeration, 33, 1142-1151